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USAAVLABS TECHNICAL REPORT 67-47

DESIGN AND DEVELOPMENT OF SMALL, SINGLE-STAGE CENTRIFUGAL COMPRESSOR (U)

By

A. B. Welliver
J. Acurio

September 1967

AD 385595

FILE COPY

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-173(T)

THE BOEING COMPANY
SEATTLE, WASHINGTON

Downgraded from [illegible] to [illegible] and will appear in [illegible] 1 March 1969 (cited in [illegible]) [illegible]

Fred J. [illegible] [illegible] January 19[illegible]

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(C) SUMMARY (U)

The research program discussed in this report was conducted for the U. S. Army Aviation Materiel Laboratories. The purpose of the work was to define, by analysis and experimental evaluation, the design criteria and performance characteristics of the high-pressure-ratio single-stage centrifugal compressor. The overall performance target was a pressure ratio of 10:1 at an adiabatic efficiency of 80 percent at an airflow of 2 pounds per second. The research was expected to lead to development of advanced technologies applicable to small gas turbine engines. The potential advances identified were:

1) Doubling of current power to weight ratio.

2) Reducing full and part load fuel consumption.

3) Minimizing cost per horsepower.

To illustrate the advances possible, two types of thermodynamic cycles (simple and regenerative) were studied. It was shown that a compressor meeting the above targets would provide an opportunity for reducing specific fuel consumptions to 0.49 pound per horsepower per hour (simple cycle) and 0.38 pound per horsepower per hour (regenerative).

RESEARCH COMPONENTS

Studies of compressor designs by the contractor in the pressure ratio range of 3.5:1 to 7:1 formed the background for initial work in this program.

Principal areas of investigation were directed toward

1) Minimizing the effect of transonic flow conditions at the inducer and diffuser entrance, including the evaluation of inducer hub-to-tip diameter ratio.

2) Establishing methods to provide operating range at high-pressure ratio.

3) Establishing flow models for the impeller and diffuser to identify losses throughout the compressor and to determine improved techniques for future compressor design. The results of the above effort are presented in Reference 1.

At the completion of the effort described in Reference 1, the goal of 10:1 pressure ratio at 80-percent adiabatic efficiency still appeared to be possible. A new impeller and diffuser were designed and tested to evaluate the concepts reported in Reference 1 and to provide additional information where necessary to meet the target. The new impeller and diffuser were designated RF-2 and V1 respectively.

DTX 312

NASA CR-54968
APS-5211-R

NASA

BRAYTON CYCLE
3.2-INCH RADIAL COMPRESSOR
PERFORMANCE EVALUATION

N66 29906	
(ACCESSION NUMBER)	(THRU)
130	1
(PAGES)	(CODE)
CR-54968	03
(NASA CR OR TMX OR AD NUMBER)	(CATEGORY)

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MP-13492

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Patented May 17, 1949 2,470,565

UNITED STATES PATENT OFFICE

2,470,565

SURGE PREVENTING DEVICE FOR CENTRIFUGAL COMPRESSORS

Isidor R. Loss, Rowayton, Conn., assignor to Ingersoll-Rand Company, New York, N. Y., a corporation of New Jersey

Application October 9, 1945, Serial No. 621,193

5 Claims. (Cl. 230—115)

This invention relates to compressors, but more particularly to compressors of the centrifugal type.

One object of the invention is to prevent the surging of compressed air back through the discharge line of the compressor into the impeller chamber.

Other objects will be in part obvious and in part pointed out hereinafter.

In the drawings accompanying this specification and in which similar reference numerals refer to similar parts,

Figure 1 is a longitudinal elevation, partly broken away, of a centrifugal compressor equipped with a surge preventing device constructed in accordance with the practice of the invention and showing the device connected for operation in response to pressure values existing at relatively spaced points in the discharge channel of the compressor, and

Figure 2 is a view similar to Figure 1 showing the surge preventing device connected for operation in response to pressure values existing at relatively spaced points in the inlet conduit of the compressor.

Referring more particularly to the drawings, 20 designates a centrifugal compressor the casing 21 of which is recessed to provide an impeller chamber 22, a diffusor chamber 23 encircling the impeller chamber and a volute discharge chamber 24 around the diffusor chamber 23. Within the impeller chamber 22 is an impeller 25 for pumping fluid medium from the inlet opening 26 of the compressor to the chambers 23 and 24, and said impeller is interlockingly secured to the end of a shaft 27 extending axially into the impeller chamber.

In accordance with the practice of the invention, the compressor is provided with means for preventing the compressed medium from breaking back through the discharge line into the impeller chamber whenever the normal pressure condition in the discharge line is disturbed, as when operating at light load or when a machine operated by the compressed fluid is suddenly shut-off and in consequence of which a pressure will build up in the discharge channels of the compressor that will overcome the current issuing from the impeller and cause a pulsation or pumping effect in the compressor.

The means serving to obviate this unfavorable occurrence includes a relief port 28 for the discharge channel and a passage 29 leading from the port 28 to the inlet opening 26. The port 28 opens into the diffusor chamber 23. It is preferably located immediately adjacent the impeller tip and is controlled by a valve 30 of the poppet type the stem 31 of which carries a piston 32 that is slidable in a cylinder 33 on the compressor casing 21.

The outer end of the cylinder is sealed by a cover 34 that also acts as an abutment for the end of the stem 31 which, in the example shown, projects from the piston 32 to abut the cover 34 and hold the piston in spaced relation therewith to define a pressure chamber 35 in the outer end of the cylinder 33 for fluid medium serving to hold the valve 30 in its port-closing position. The inner end of the cylinder 33 contains a compression spring 19 that seats against the end wall of the cylinder 33 and against the piston 32 and is biased to hold the valve 30 away from the port 28.

Any suitable pressure medium may be introduced into the chamber 35 for holding the valve 30 in position to seal the port 28, and such medium is conveyed to and from the chamber 35 by a conduit 36 leading from a valve casing 37 that may be affixed to the compressor and has a chamber 38 for the accommodation of a reciprocatory valve 39. The valve 39 is of generally cylindrical shape and in its periphery is an annular groove 40 to afford communication between the conduit 36 and a pressure medium supply conduit 41 threadedly connected to the casing 37.

The valve 39 also controls the exhaust of fluid medium from the chamber 35 and to this end is provided, in the side thereof, with a recess 42 of suitable length to span the end of the conduit 36 and an end of a discharge conduit 43 threaded into the casing 37 and communicating with the chamber 38. The ends of the casing 37 are suitably vented to the atmosphere to assure the unrestricted movement of the valve 39 in the casing.

The valve 39 is shifted by an actuator that moves responsively to a differential existing between pressures acting on the opposed sides thereof and derived from up-stream and down-stream points in the discharge channel of the compressor. In the form illustrated the actuator consists of a flexible diaphragm 44 the marginal portion of which is clamped between two plates 45 and 46 that are recessed to provide pressure chambers 47 and 48, respectively.

The axial portion of the diaphragm 44 is secured to the end of a stem 49 carried by the valve 39, and the aperture 50 in the plate 46 through which the stem 49 extends is sealed by a suitable packing device 51. The diaphragm 44 is so positioned with respect to the valve 39 that when the

REMAND DTX 313

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diaphragm occupies the neutral or relaxed position both the annular groove 40 and the recess 42 will be out of communication with the supply conduit 41 and said conduit will be blanked off by a cylindrical portion of the valve 39 located between the annular groove 40 and the recess 42.

The pressure fluid serving to actuate the diaphragm is conveyed to the chamber 48 from the diffusor chamber 23 by a conduit 52 threadedly connected to the cover 45 at one end and at its other end to the casing 21 to communicate with a port 53 opening into the diffusor chamber 23 adjacent the tip 54 of the impeller 25. Similarly, a conduit 55 threadedly connected to the cover 45 and to the casing 21 conveys pressure fluid from the discharge end 56 of the volute chamber 24 to the chamber 47.

During the normal operation of the machine, the superior pressure in the chamber 47 acting against the diaphragm will shift the valve 39 toward the right hand end of the casing 37 and thereby place the annular groove 40 in communication with the supply conduit 41 and the conduit 36. Pressure medium will then flow through the conduits into the chamber 35 against the piston 32 and move the valve 30 into sealing relationship with the port 28. All the air thereafter discharged by the impeller will pass through the discharge channels of the compressor, for utilization, and the valves 39 and 30 will remain in the positions described as long as the pressure in the chamber 47 predominates over that in the chamber 48.

If, for any reason, the pressure in the discharge channel surges back toward the impeller, such impulse will be communicated through the conduit 52 against the side of the diaphragm in the chamber 48 and will shift the valve 39 leftward. In this position the valve 39 will blank off the supply conduit 41 and will establish communication between the conduits 36 and 43 to effect an outlet for the pressure medium in the chamber 35. The spring 19 will then shift the valve 30 away from the port 28 and permit the compressed fluid surging in the direction of the impeller to escape through the port 28 and the passage 29 into the inlet passage 26.

The valves 39 and 30 will occupy these positions only until pressure conditions in the discharge channel of the compressor again become normal, whereupon the normal pressure differential will again be established in the chambers 48 and 47 and the valve 39 will then again be shifted to effect closing of the valve 30.

In the modified form of the invention shown in Figure 2, the diaphragm 44 and the valve 39 move in response to a differential between pressures existing at relatively spaced points in the inlet conduit 57 of the compressor which has means for creating a sub-normal pressure therein. Such means is shown as being in the form of an orifice meter 58 and the space encircling it and that directly adjacent thereto constitute a zone 59 of sub-normal pressure. This zone 59 is in communication with the chamber 48 through a conduit 60, and a conduit 61 affords communication between the chamber 47 and the inlet conduit 57 at a point on the up-stream side of the orifice meter 58.

In this form of the invention, a spring 62 is interposed between the plate 46 and the diaphragm 44 to augment the pressure in the chamber 48 for shifting the valve 39 leftward to blank off the supply conduit 41 and to place the valve

39 in position to establish communication between the conduits 36 and 43.

During the operation of the device, the pressure within the chamber 47, being normal inlet pressure, will overcome the pressure of the fluid in the chamber 48 and that of the spring 62 and move the valve 39 to the right hand limiting position in the chamber 38. In this position of the valve 39 pressure medium will flow into the chamber 35 against the piston 32 and move the valve 30 into sealing relationship with the port 28.

The valves will remain in the positions described as long as pressure conditions in the inlet and discharge channels remain normal. But if the pressure in the discharge channel closely approaches the value at which it may surge back toward the impeller this pressure condition is communicated through the impeller chamber and the zone 59 to the chamber 48 and will cause the pressure therein, together with the force exerted by the spring 62 to shift the valve 39 leftward.

During this movement, the valve will first cut-off the pressure chamber 35 from the source of fluid medium supply and next place said chamber into communication with the atmosphere through the recess 42 and the conduit 43. The spring 19 will then shift the valve 30 away from the port 28, and any compressed fluid surging back through the discharge channel in the direction of the impeller will escape through the port 28 and the passage 29 into the inlet conduit. These movements of the valves will take place in anticipation of a surge in the discharge line and the port 28 will, therefore, be uncovered before the surge reaches the impeller.

I claim:

1. In combination, a centrifugal compressor having an inlet channel and a discharge channel, an impeller to pump fluid from the inlet channel into the discharge channel, a relief port for the discharge channel adjacent the tip of the impeller, and means acting in response to an abnormal ratio of pressure values substantially at the impeller and at a point existing at a point relatively spaced from the first mentioned point to valve fluid from the discharge channel and thereby prevent the surging of compressed fluid from the discharge channel back through the impeller.

2. In combination, a centrifugal compressor having an inlet opening and a discharge channel and a diffusor chamber, an impeller to pump fluid from the inlet opening into said channel, a relief port for the diffusor chamber adjacent the tip of the impeller, and means acting in response to an abnormal ratio of fluid pressure values existing at a point substantially at the tip of the impeller and at a point relatively spaced from the first mentioned point to valve fluid from said discharge channel and thereby prevent the surging of compressed fluid from the channel back through the impeller.

3. In combination, a centrifugal compressor having an inlet opening and a discharge channel, an impeller to pump fluid from the inlet opening into the discharge channel, a relief port positioned adjacent the tip of the impeller for the discharge channel, a fluid actuated relief valve to govern fluid flow through the relief port, a spring acting constantly to position the relief valve for opening the port, a control valve for regulating flow of a pressure medium to position the relief valve for closing the port and to control the exhaust of such pressure medium from the relief valve, and a flexible diaphragm acting respon-

sively to a pressure differential existing in the discharge channel for actuating the control valve.

4. In combination, a centrifugal compressor having an inlet opening and a discharge channel and a diffusor chamber, an impeller to pump fluid from the inlet opening into the discharge channel, a relief port for the diffusor chamber positioned adjacent the tip of the impeller, a fluid actuated relief valve for the relief port, a control valve to regulate the flow of a pressure fluid to the relief valve and to exhaust said fluid therefrom, means within the inlet channel for creating a subnormal pressure area therein, and a flexible diaphragm responsive to pressure variations in the subnormal pressure area for actuating the control valve.

5. In combination, a centrifugal compressor having inlet and discharge channels, an impeller to pump fluid from the inlet channel into the discharge channel, a relief port for the discharge channel positioned adjacent the tip of the impeller, a fluid actuated relief valve to regulate

flow through the relief port, a spring acting constantly to position the relief valve for opening the port, a control valve for regulating flow of a pressure medium to position the relief valve for closing the port and for controlling the exhaust of such pressure medium from the relief valve, and a flexible diaphragm for actuating the control valve acting responsively to a pressure differential existing in one of said channels.

ISIDOR R. LOSS.

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2,342,219	Price	Feb. 22, 1944

May 17, 1949. I. R. LOSS 2,470,565

SURGE PREVENTING DEVICE FOR CENTRIFUGAL COMPRESSORS

Filed Oct. 9, 1945

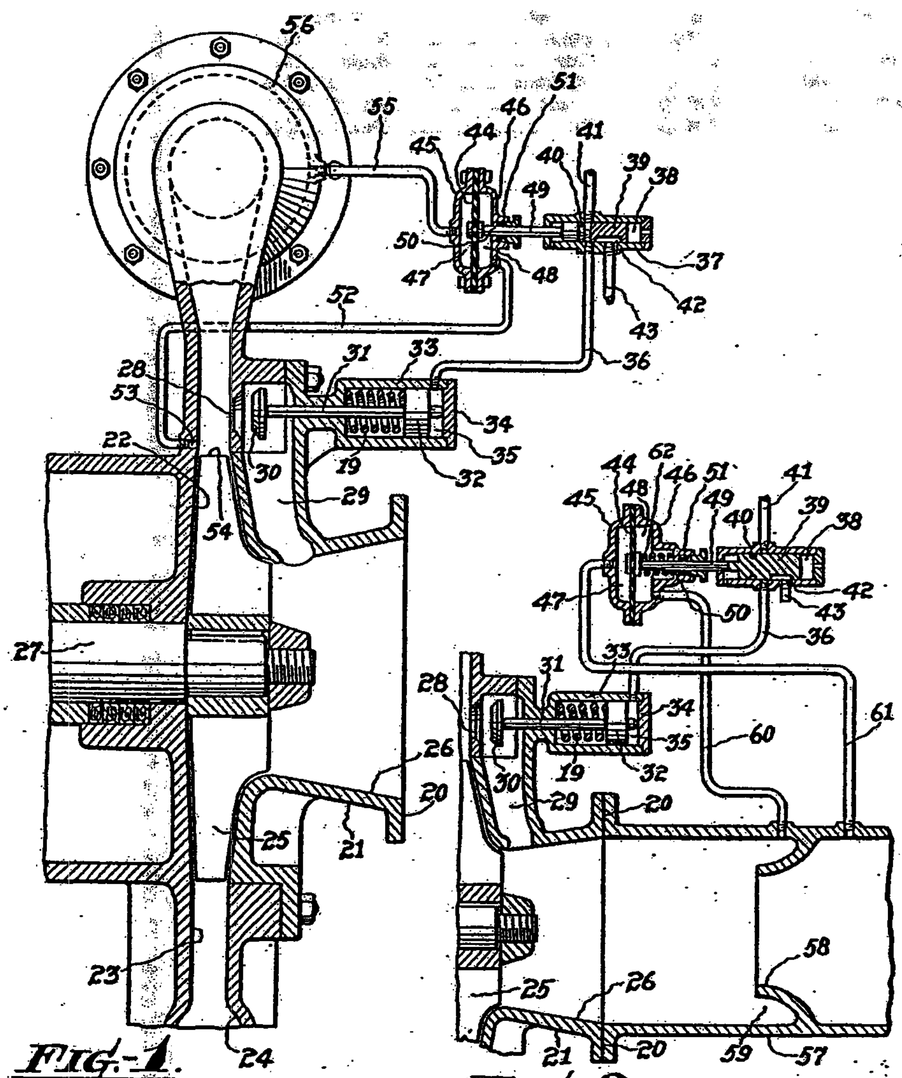

FIG-1.

FIG-2.

INVENTOR
Isidor R. Loss.

BY [signature]
HIS ATTORNEY.

DTX 317

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,047,210

July 31, 1962

Stanley G. Best

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 3, line 1, the equation should appear as shown below instead of as in the patent:

$$\Delta P=K(P_2-P_1)$$

column 6, line 3, for "large" read -- larger --.

Signed and sealed this 11th day of December 1962.

(SEAL)
Attest:

ERNEST W. SWIDER
Attesting Officer

DAVID L. LADD
Commissioner of Patents

REMAND
DTX 317

DTX 327

United States Patent [19] [11] 4,164,033
Glennon et al. [45] Aug. 7, 1979

[54] COMPRESSOR SURGE CONTROL WITH AIRFLOW MEASUREMENT

[75] Inventors: Timothy F. Glennon; Theodore E. Sarphie; Dennis T. Faulkner, all of Rockford, Ill.

[73] Assignee: Sundstrand Corporation, Rockford, Ill.

[21] Appl. No.: 833,031

[22] Filed: Sep. 14, 1977

[51] Int. Cl.² F04D 27/02; F02C 9/14
[52] U.S. Cl. 364/431; 60/39.29; 415/17; 415/39
[58] Field of Search .... 364/431; 340/27 SS; 415/27, 28, 17, 39; 73/115–117

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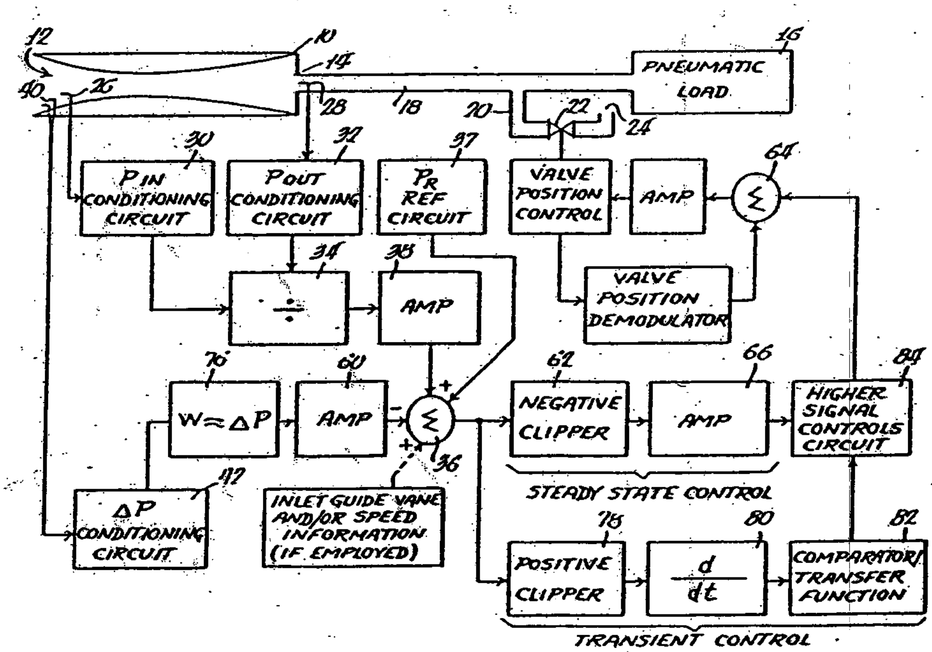
Primary Examiner—Felix D. Gruber
Attorney, Agent, or Firm—Wegner, Stellman, McCord, Wiles & Wood

[57] ABSTRACT

Surge control systems are provided for compressors which supply air to a pneumatic load. A signal proportional to the pressure ratio of the outlet pressure to the inlet pressure plus a selected reference pressure is compared to a measured weight flow rate of air through the compressor to provide a vent valve command signal. If the measured pressure ratio plus the reference pressure ratio exceeds the measured flow rate, a surge condition may ensue. The vent valve position command signal causes a venting valve to vent a portion of the air provided to the load, which reduces the measured pressure ratio and increases weight flow rate. During normal operation of the compressor, the vent valve command signal is zero for all values of weight flow rate. A transient control channel is provided which is responsive to the rate of change of the measured pressure ratio and/or measured weight flow rate.

11 Claims, 3 Drawing Figures



example, to the right of the surge line. It is desirable that the system maintain a pressure ratio P_r equal to or greater than the P_r value at the intersection of the operating line with the common speed line (or inlet guide vane position line) but less than the P_r value at the intersection of the surge line and the common speed line.

In the present invention, the pressure ratio P_r is controlled by a venting valve which increases or decreases the output pressure P_{out} and weight flow rate W so that P_r equals the P_r value at the intersection of the common speed line with the operating line. The venting valve is controlled when the P_r value is in the surge correction region as shown in FIG. 1. The position of the valve determines the value of P_r and W and is controlled by a surge control circuit to be explained in greater detail below.

If the compressor is operating in surge condition (i.e., on the surge line), the valve is fully opened to most rapidly reduce P_r and increase W. If the compressor is operating in the normal operating region (i.e., on the operating line), the valve is fully closed. As the venting valve is opened, the pressure ratio P_r drops and the weight flow rate W increases along the common speed line (or along the common IGV line) toward the point of intersection with the normal operating line. As the pressure ratio approaches a value representing the normal operating line, the surge control circuit of the present invention proportionally closes the valve and completely closes it when the pressure ratio P_r lies at the intersection of the operating line. Thereafter, if the pressure ratio increases to enter the surge correction region, the control valve is opened in an amount proportional to the magnitude of the correction required to drop the pressure ratio P_r back toward the intersection with the operating line.

An explanation of the operation of various control systems for the compressors will now be provided with particular reference to a centrifugal compressor having a backward curved impeller which has an extended choke to stall range and within that range an appreciable zone of constant pressure variable flow. Although the centrifugal compressor will be described in combination with the control circuits, it should be understood that the control circuits of the present invention are capable of controlling surge for any type of compressor having a surge map similar to that shown in FIG. 1.

Referring to FIG. 2, a surge control system for a fixed speed, fixed geometry compressor is shown. A compressor 10 has an inlet 12 and an outlet 14 which supplies compressed air to pneumatic load 16 by a pneumatic conduit 18 which is coupled between the load 16 and the outlet 14. A venting conduit 20 is coupled in parallel with load 16 and has a dump valve 22 therein. The position of valve 22 determines the amount of airflow from outlet 14 to a vent 24.

A pressure sensor 26, which may be a conventional transducer or a strain gauge, measures the pressure at inlet 12 and converts it to a signal representative of the amplitude of the pressure at that point. Similarly, a sensor 28 measures the pressure at outlet 14 and provides a signal P_{out} proportional to its magnitude. The signals representing P_{in} and P_{out} are applied to conditioning circuits 30 and 32, respectively. The conditioning circuits remove noise and transients from the signals. The signals are then applied to a divider circuit 34 to divide the signal representing the outlet pressure P_{out} by a signal representing the input pressure P_{in}. The output from divider circuit 34, P_r, is applied to a sum-

mer 36 through an amplifier 38. The selection of the gain of amplifier 38 will be discussed in greater detail below.

The partially corrected weight flow W' can be expressed in the form of an equation as follows:

$$W' = c\sqrt{\Delta p\left(\frac{P_{in}}{T_{in}}\right)} \qquad \text{(EQ 1)}$$

where c is a selected airflow constant, Δp is the pressure difference at the outlet of the compressor, P_{in} is the inlet pressure, and T_{in} is the inlet temperature. Also, W' can be corrected for temperature and pressure by multiplying it by the $\sqrt{\theta/\delta}$ to equal $W'\sqrt{\theta/\delta}$ wherein θ is equal to $T_{in}/519.7°$ K. (EQ 2) and δ is $P_{in}/14.7$ (EQ 3). The multiplication of W' by the correction values assures that a more accurate weight flow rate is obtained.

Returning to FIG. 2, a sensor 40, located in outlet 14, senses Δp. The Δp signal is provided to a Δp conditioning circuit 42 to remove noise. Also, temperature sensor 44 located at the inlet 12 senses the temperature and generates a signal proportional to it which is applied to temperature conditioning circuit 46. W' calculation circuit 48 receives the signals representing temperature, Δp and input pressure P_{in} from conditioning circuits 46, 42 and 30, respectively. The circuit manipulates C, Δp, P_{in} and T_{in} to provide an output representing W' as in Equation 1, above.

Temperature and pressure correction of W' will now be considered. Temperature correction circuit 50 multiplies the signal received from temperature conditioning circuit 46 by an amount equal to that shown in Equation 2. The output from temperature correction circuit 50 is applied to a square root circuit 52 which obtains the square root of the value of the signal from the temperature correction circuit 50. The value from the square root circuit 52 is multiplied by W' by multiplier 54. The product therefrom is provided to divide circuit 56. Also provided to divide circuit 56 is the signal representing δ from pressure correction circuit 58. The output from pressure correction circuit 58 is represented by Equation 3. The output from divide circuit 56 is applied to summer 36 through an amplifier 60. The selection of the gain of amplifier 60 will be discussed in greater detail below.

A signal representing P_r ref is provided by P_r ref circuit 37 and applied to algebraic amplifier or summer 36. The signal from summer 36 is the sum of the negative signal from amplifier 60, the positive signal from amplifier 38 and the positive signal P_r ref from circuit 37. The signal from summer 36 will be hereinafter referred to as the vent valve command signal and may be expressed in the form of an equation as

$$P_r + P_r\,\text{ref} - \frac{W'\sqrt{\theta}}{\delta} \qquad \text{(EQ 4)}$$

The polarity of the signal is indicative of whether or not the system is operating in the surge correction region or in the normal operating region about a selected reference pressure P_r as shown in FIG. 1. That is to say, if the vent valve command signal is positive, the magnitude of the P_r term exceeds the magnitude of the W' term at a selected reference pressure P_r ref, and the operation of the compressor is operating in the surge

DTX 349

IN THE UNITED STATES DISTRICT COURT
FOR THE DISTRICT OF DELAWARE

HONEYWELL INTERNATIONAL INC. and HONEYWELL INTELLECTUAL PROPERTIES INC.,	)	
Plaintiffs,	)	Civil Action No. 99-309 (GMS)
v.	)	
HAMILTON SUNDSTRAND CORPORATION,	)	
Defendant.	)	

DECLARATION OF GERARD MULLER IN SUPPORT OF HONEYWELL'S RESPONSES TO SUNDSTRAND'S SUMMARY JUDGMENT MOTIONS FILED AUGUST 7, 2000

1. My name is Gerard Muller. Since 1986, I have been the President, owner and sole employee of Serry-Tech, Inc., which provides consulting services in the area of process machinery reliability assessment, detail compression and power machinery assessments and selection for new projects, and compressor and turbine technology training. The machinery involved includes flight derivative gas turbines for compressor drives, high and low speed centrifugal process compressors, high speed centrifugal air compressors and the evaluation of their associated operating and surge controls. I have a Masters degree in Mechanical Engineering and more than 35 years of experience in working with the design and control of compressors. My background and experience are outlined in the resume attached to this affidavit.

2. In the 1970s and through the mid-1980s, I was working for Exxon Research & Engineering Company, where my responsibilities included evaluating gas turbines, steam turbines, turbo-expanders and axial and centrifugal compressors for new projects, as well as troubleshooting this family of equipment in plants worldwide. Troubleshooting included evaluating the mechanical and operational performance of these power and compression services

REMAND
DTX 349

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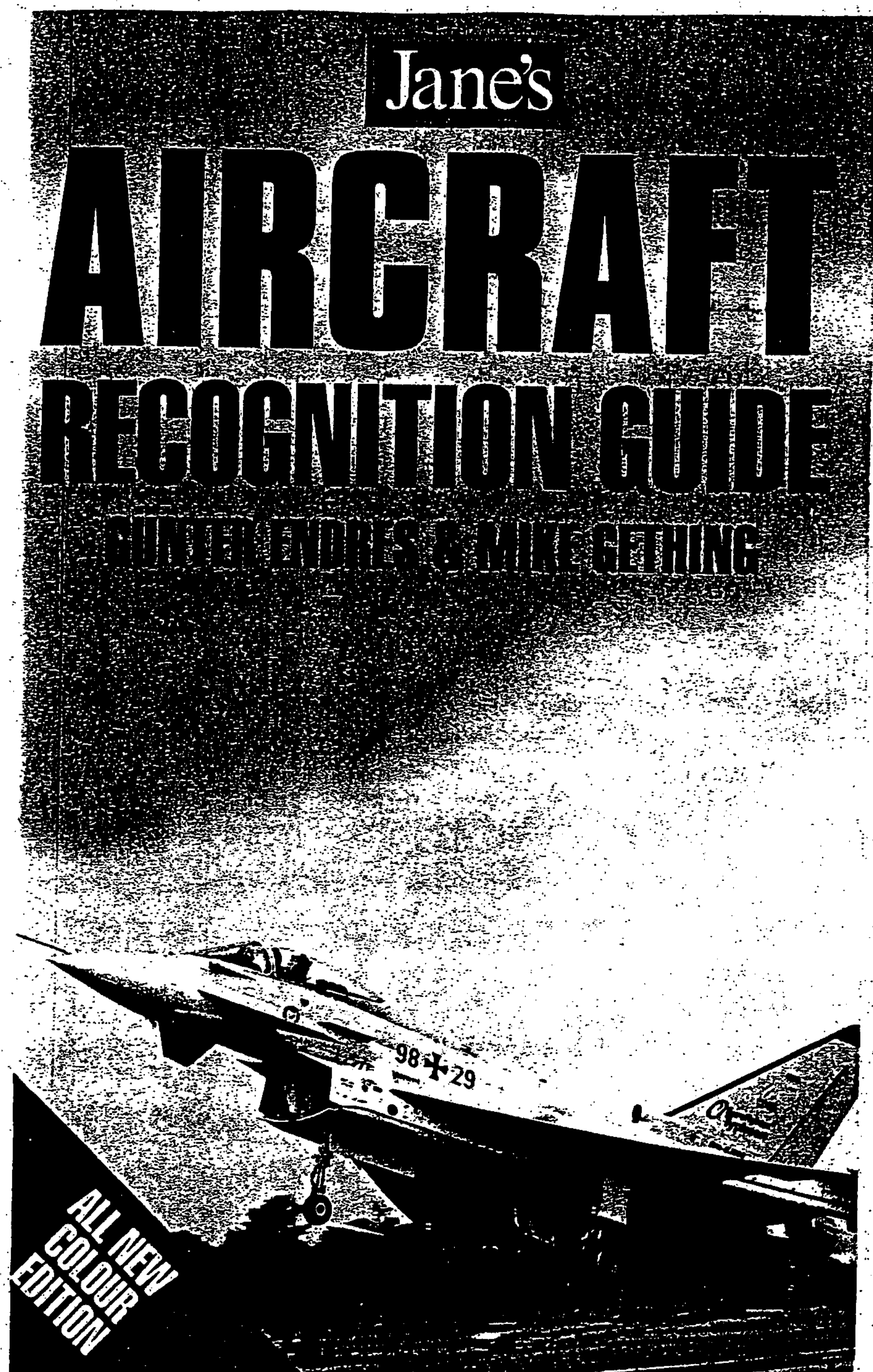
point in the surge control process. As stated by Sundstrand in its brief, the APS 3200 measures the position of the IGVs as part of the test to determine whether the system is in "high-flow" or "low-flow" mode. (HS '893/'194 Brf. at 17) IGV position therefore serves as a set point for the "low-flow"/"high-flow" determination. That is, if the IGV position is below a certain value, it indicates that the system is operating in "low-flow" mode. (Szillat Dep. Tr. at 152-53) When in "low-flow" mode, the system uses the flow-related parameter and proportional and integral controllers to control the surge bleed valve. (Suttie Decl. at ¶17; HS '893/'194 Brf. at 17) Therefore, the APS 3200 uses the position of the IGVs in a similar way as what is described in the '893 and '194 patents. The result of using the IGV position in the APS 3200 is the same as the result described in the patent – the system responds to different IGV positions in controlling the surge bleed valve. So the use of IGV position in the APS 3200 is equivalent to the use described in the '893 and '194 patents.

34. Sundstrand attempts to distinguish the APS 3200's use of IGV position by discussing the "inverted-V/double solution" characteristic of the APS 3200. (HS '893/'194 Brf. at 16-17) But that "inverted-V/double solution" characteristic has more to do with where pressure is being measured in the compressor than how the APS 3200 controls surge. The high performance, high speed, centrifugal air compressors (i.e., load compressors) utilized in APUs commonly produce supersonic conditions in the diffuser (a part of the load compressor). (See Shapiro, The Dynamics and Thermodynamics of Compressible Fluid Flow, The Ronald Press Co., 1953, pp. 73-141). Supersonic refers to air velocity that is greater than the speed of sound. As the velocity of the air flow reaches supersonic levels, it causes a shock wave to travel through portions of the compressor. A shock wave produces large, almost instantaneous, pressure changes. During operation of the APS 3200, the air flow reaches supersonic rates. As part of the

flow-related parameter that measures air flow in the APS 3200, static pressure is measured in the diffuser. (Suttie Dep. Tr. at 239) Before the shock wave reaches that static pressure tap in the APS 3200, the static pressure increases. Once the shock wave passes the pressure tap, the static pressure decreases. Thus, the "inverted-V/double solution" characteristic is a result of the shock wave passing by the static pressure tap in the diffuser. The existence of the "inverted-V/double solution" characteristic in the APS 3200 therefore has nothing to do with whether or not the APS 3200 uses the technology in the '893 and '194 patents. Instead, the "inverted-V/double solution" characteristic is strictly a result of the location of the static pressure tap. Any compressor taking a static pressure measurement of supersonic air flow in the diffuser would have a similar characteristic. Sundstrand's argument relating to the "inverted-V/double solution" characteristic in the APS 3200 therefore does not negate the fact that the APS 3200 uses the technology described in the '893 and '194 patents.

35. In addition to doing the equivalent of many of the claims of the '893 and '194 patents, the APS 3200 literally includes all of the elements of claim 4 of the '194 patent. Element (a) of claim 4 requires a supply duct that connects the compressor with the pneumatic system of the aircraft. ('194 Pat., Col. 11) Sundstrand admits that the APS 3200 has such a supply duct. (RFA24; '194 Cl. Cht. at 1) Element (b) of claim 4 requires the discharge of air from the compressor, through the supply duct, to the pneumatic system. ('194 Pat., Col.11) Sundstrand admits that the APS 3200 discharges air from the compressor through a supply duct to a pneumatically-powered apparatus. (RFA24; '194 Cl. Cht. at 2) Element (c) of claim 4 requires that the APU use a flow-related parameter to measure air flow through the supply duct, use proportional and integral controllers to generate control signals in relation to the value of the flow-related parameter, and simultaneously use those control signals to control a surge bleed

DTX 376



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Jane's

AIRCRAFT

RECOGNITION GUIDE

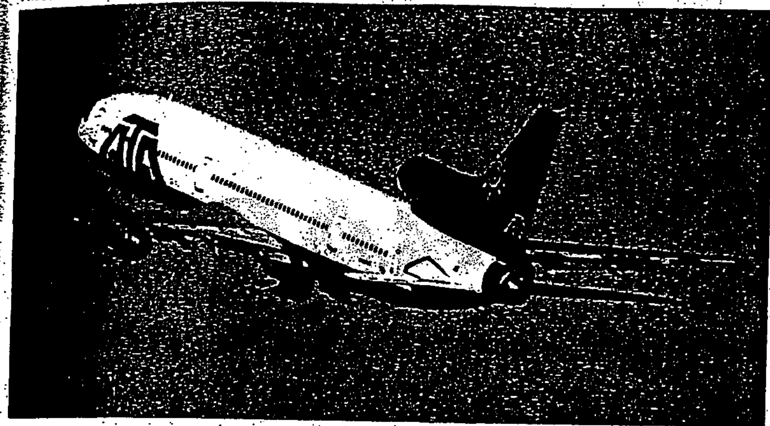
THIRD EDITION

Compiled and Edited
by
Günter Endres
and
Michael J. Gething

HarperCollins*Publishers*

Lockheed L1011 TriStar USA

Three-turbofan medium-haul airliner



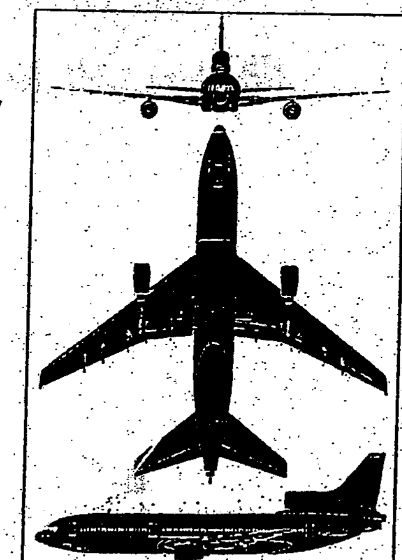
Developed to an American Airlines requirement for a high capacity aircraft with transcontinental range and able to take off from comparatively short runways. First flown on 17 November 1970, the TriStar entered revenue service with Eastern Air Lines on 26 April 1972. Total delivered: 250.

FEATURES

Low/swept wings; three R-R RB 211 turbofans, two on pylons under wing, the third at base of and integrated with swept tailfin; low-set swept tailplane; widebody fuselage

VARIANTS

L1011-1: Initial production version
L1011-50: Conversion of TriStar 1 with higher operating weight
L1011-100: Higher gross weight and fuel capacity
L1011-150: Conversion of TriStar 1 to increase range capability
L1011-200: Uprated engines and higher gross weight
L1011-250: Converted TriStar 1 with same engines as Model 500
L1011-500: Shorter fuselage, long-range with aerodynamic improvements



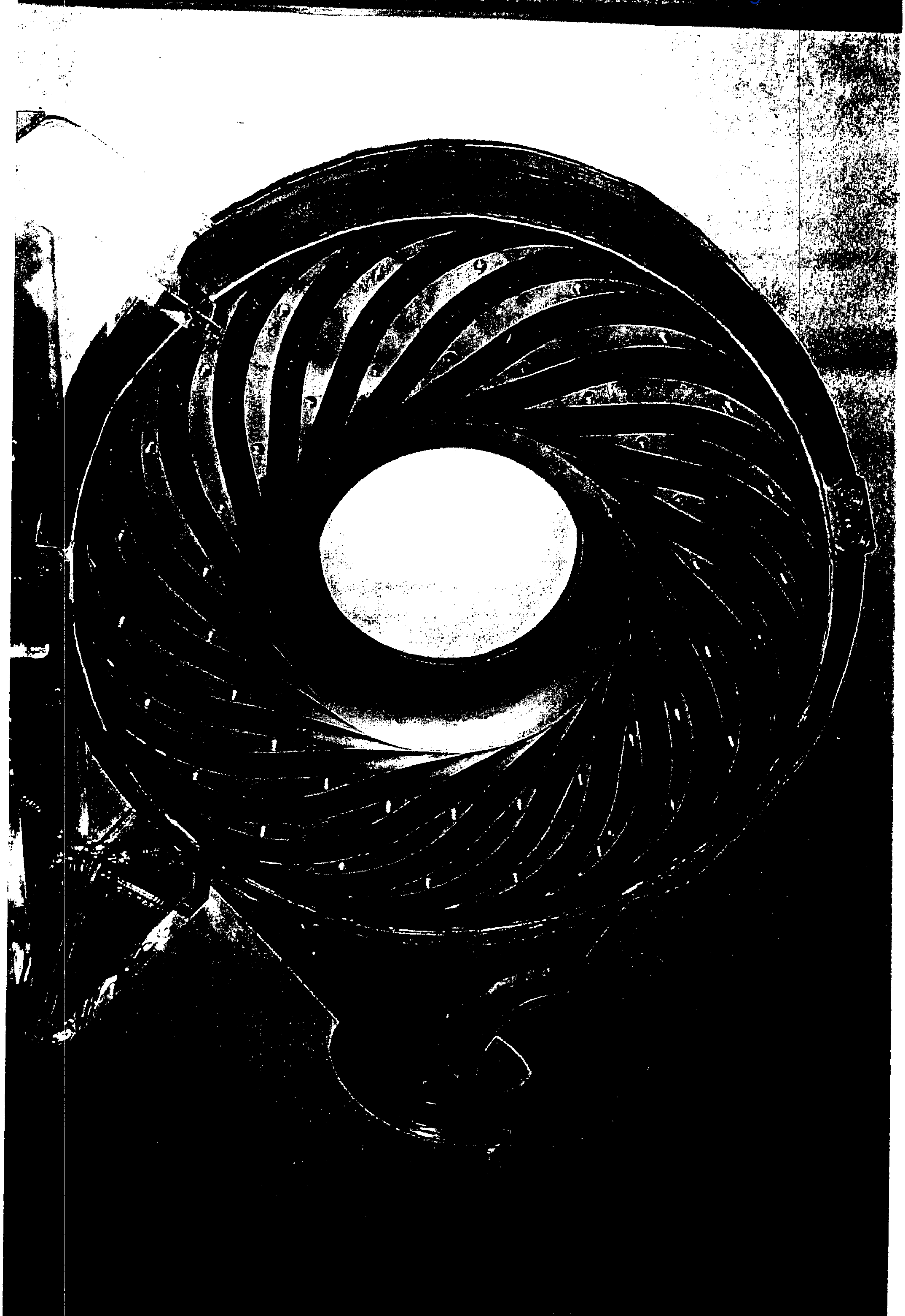
SPECIFICATIONS: L1011-500

Accommodation: 3 + 330
Cargo/baggage: 118.9 m^3 (4,200 cu.ft)
Max speed: M0.84 (481 kt; 890 km/h)
Range: 5,297 nm (9,815 km)

DIMENSIONS

Wingspan: 47.3 m (155 ft 4 in)
Length: 50.1 m (164 ft 3 in)
Height: 16.9 m (55 ft 4 in)

DTX 399



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